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DESIGN AND DEVELOPMENT OF LIQUID HYDROGEN COOLED 120MM ROLLER, 110MM ROLLER, AND 110MM TANDEM BALL BEARINGS FOR M-1 FUEL TURBOPUMP

By

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National Aeronautics and Space Administration

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AEROJET-GENERAL CORPORATION

SACRAMENTO, CALIFORNIA

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TECHNOLOGY REPORT

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ABSTRACT

The results of a liquid hydrogen bearing development program for the M-1 liquid hydrogen turbopump are presented in this report. Roller bearings of 110 mm diameter were loaded to 5000 lb at 13,300 rpm for 2770 sec with a coolant rate of 50 gpm. A triple ball bearing set, 110 mm diameter, was loaded to 36,000 lb at 13,300 rpm for 2700 sec with a coolant rate of 150 gpm. Four roller bearings, 120 mm diameter, were loaded to 15,500 lb at 13,300 rpm for 5880 sec with a coolant rate of 26 gpm. Acceleration tests were successful with rates of 28,000 rpm per second for the 120 mm diameter roller bearings and 15,000 rpm per second for the 110 mm diameter ball bearings. DN values were in the range of 1.6×10^6.

author

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I. SUMMARY

The results of the bearing development program, conducted for the M-1 liquid hydrogen turbopump during 1964 and 1965, are delineated in this report. Design expectations were confirmed with no significant design changes. Where a bearing failure occurred, the cause was either tester misalignment or inadvertent low coolant flow.

The successful operation of bearings using liquid hydrogen from the pump stream eliminates the need for a separate bearing coolant system. The main disadvantage of this type of coolant supply is low pressure and consequent low flow at the start-up of the turbopump.

The design operating requirements were as follows:

		Pump Roller	Turbine Roller	Tandem Ball
Load	(lb)	4,600	10,800	35,000
Speed	(rpm)	13,300	13,300	13,300
Acceleration	(rpm/sec)	14,400	14,400	14,400

The low coolant flows and pressures, produced at low turbopump speeds, were used as low limit criteria for peripheral testing. When the coolant flows were reduced below these limits, bearing failure occurred.

Four of the 120 mm roller bearings accumulated in excess of 5000 sec testing time at rated speed and at 40% overload. Two of the 110 mm triple ball bearing sets accumulated 2700 sec at rated speed and load. Acceleration tests were conducted up to 28,000 rpm/sec on roller bearings and 15,000 rpm/sec on ball bearings. The load sharing tests, performed on the triple ball bearing set, received special attention because good load sharing prolongs safe turbopump operation.

II. INTRODUCTION

The results of the bearing development program conducted by the Aerojet-General Corporation under Contract NAS3-2555 to the National Aeronautics and Space Administration, Cleveland, Ohio, are delineated herein. The objective of the program was to evaluate the performance of liquid hydrogen cooled bearings under simulated turbopump conditions of loads, speeds, pressures, and accelerations. The arrangement of the bearings in the turbopump and in the testers is also discussed as are the test procedures and an analysis of the results for ball and roller bearings.

A similar program to evaluate the performance of ball and roller bearings in liquid oxygen was also conducted and the results are reported independently.[1] It was indicated from the results of the liquid oxygen bearing development program that the testing of a large number of bearing configurations was unnecessary for the selection of a bearing design suitable for the intended loads, speeds, and accelerations. Previous tests performed using commercially-available thrust bearings yielded valuable operational information, such as pre-test cooling requirements, load and speed control relationships, and tester capability.

The liquid hydrogen cooled bearing test program concentrated upon the evaluation of a single configuration of 110 mm ball bearings and 110 mm roller bearings manufactured by Industrial Tectonics Inc., and 120 mm roller bearings manufactured by the Bower Roller Bearing Division of Federal Mogul.

III. DESCRIPTION AND INSTALLATION

A. TURBOPUMP

The M-1 fuel turbopump is a multistage axial flow pump driven by a directly connected gas turbine. The function of the fuel turbopump is to provide high pressure liquid hydrogen to the M-1 rocket engine. The rotating parts of the M-1 fuel turbopump (see Figure 1) are supported by two roller bearings and three ball bearings. The roller bearings are located at either end of the turbopump on 42.23-in. centers. The pump-end roller bearing measurements are 110 mm inside diameter x 150 mm outside diameter x 20 mm width. The turbine-end roller bearing measurements are 120 mm inside diameter x 180 mm outside diameter x 28 mm width.

The operating conditions for the roller bearings were:

	Pump Roller	Turbine Roller
Radial Load, lb	4,600	10,800
Overload, lb	5,100	11,900
Speed, rpm	13,300	13,300
Overspeed, rpm	15,000	15,000
Acceleration Rate, rpm/sec	14,400	14,400
Coolant Temperature, °F	-420	-420
Coolant Medium	LH_2	LH_2

The three ball bearings make a matched set with each bearing having a 110 mm inside diameter x 170 mm outside diameter x 28 mm width. The set is ground for load sharing with the thrust load in one direction. The estimated percentage of load sharing is 40, 30, and 30. The ball bearing set is located near the pump-end

(1) Young, M. W. and Kirby, L. F., Development of Liquid Oxygen Cooled 110 mm Roller and Tandem Ball Bearings at up to .5 x 10^6 DN Values for the Oxidizer Turbopump of the M-1 Engine, NASA CR 54816 (AGC 8800-23), dated 28 February 1966

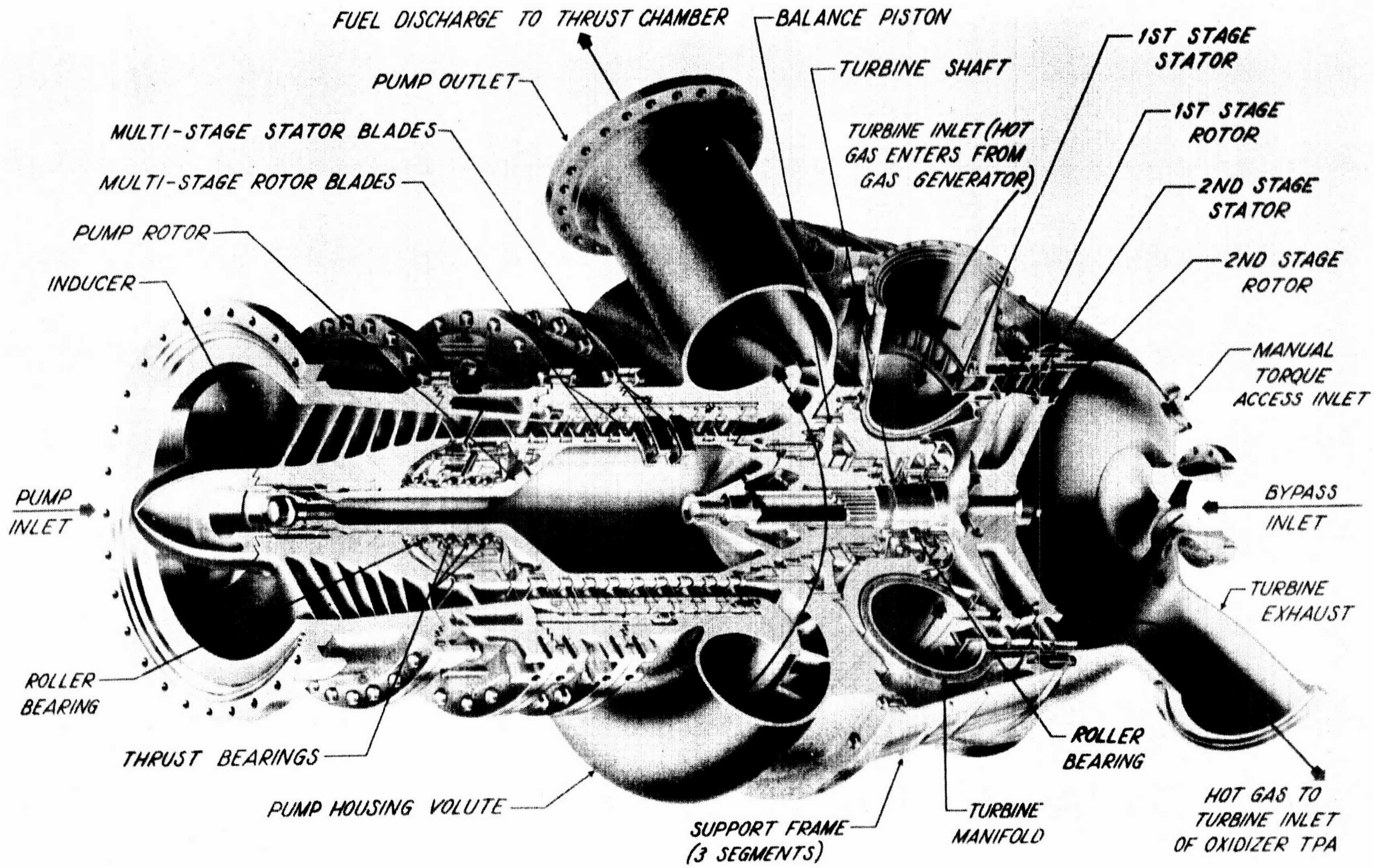

Figure 1

M-1 Turbopump Assembly

roller bearing and is mounted in a radially flexible housing to limit radial load to approximately 200 lb. The mechanical design of the flexible housing is discussed as part of the M-1 axial flow liquid hydrogen pump report.(2)

The operating conditions for the tandem thrust bearings were as follows:

Thrust Load (Sharing Direction), lb	35,000
Thrust Load (Reverse Direction), lb	6,400
Overload (Sharing Direction), lb	50,000
Speed, rpm	13,300
Overspeed, rpm	15,000
Acceleration Rate, rpm/sec	14,400
Coolant Temperature, °F	-420
Coolant Medium	LH_2

To eliminate the need for a separate bearing cooling system, all bearings were designed for liquid hydrogen cooling from a diverted portion of the pumped fluid. The coolant is directed on the rolling elements by jets with the discharge from each bearing being returned to the pump fluid stream. This parallel system gives all bearings the benefit of -420°F liquid hydrogen.

The material for the housings and shafts is Inconel 718 with the exception of the ball bearing housing which is K-Monel, and the turbine-end roller bearing shaft which is Rene' 41. The room temperature dimensions of mating housings and shafts are calculated so that the proper fits and clearances would be obtained when operating at a temperature of -420°F and a speed of 13,300 rpm.

Bearing fits and internal clearances when operating at 13,325 rpm and -420°F are as follows:

Designation	Part No.	Internal Clearance Dia.(in.)	Inner Race to Shaft	Outer Race to Housing	Axial Clearance(in)
Pump-End Roller	288260	0.0017 0.0022 Loose	0.0000 0.0007 Tight	0.0007 0.0016 Loose	-- -- --
Turbine-End Roller	288340	0.0008 0.0017 Loose	0.0002 0.0008 Tight	0.0003 0.0021 Loose	-- -- --
Thrust Ball	288410	0.0031 0.0051 Loose	0.00065 0.0012 Tight	0.0005 0.0015 Tight	0.0064 0.0014 Loose

(2) Regan, P. J., (u) Mechanical Design of the M-1 Axial Flow Liquid Hydrogen Pump, NASA CR-54823 (AGC 8800-18), 15 February 1966 (Confidential)

Based upon the following total contraction coefficient from ambient to -420°F

		Variation in Coefficient*
440C	0.0020 in./in.	0.0019/0.0023 in./in.
Rene' 41	0.0022 in./in.	0.0022/0.0026 in./in.
Inconel 718	0.00273 in./in.	0.0023/0.0027 in./in.
K-Monel	0.00245 in./in.	Not Tested

NOTE: This variation was determined by a literature search and measurements by independent laboratories. A difference in coefficient of 0.0004-in. for 120 mm bearing = 0.00189-in. difference at the inside diameter.

Bearing races, balls, rollers, and coolant rings are fabricated of 440C material, heat treated to Rockwell C-60. A typical coolant spray ring with sixteen 0.080-in. discharge holes and sixteen 0.125-in. return holes is shown in Figure 2. Sealing between inlet and return passages is accomplished by outer diameter interference fits.

B. BEARING TESTERS

1. Motor-Driven Tester (P/N 299102 and P/N 299103)

The basic configuration of the test head consists of a housing, load appliers, mounting provisions, connectors, and test cartridges. These testers are shown in Figures 3, 4, and 5. Figure 3 depicts the radial bearing test head and Figure 4 shows the tandem thrust bearing configuration. The test cartridge includes the test bearings, shaft, and mountings. The test head can be driven by either an electric motor or a turbine and it can be mounted in either the vertical or horizontal position. This test head is designed for operation at speeds up to 20,000 rpm with constant angular acceleration from 0 to 16,000 rpm in 0.8 sec. Maximum deceleration from 20,000 rpm to stop is 4 sec. All loads applied to bearings are reproducible within ± 5%. The thrust load is controllable in increments of 1000 lb (±200), from no load to 25,000 lb per bearing or a total of 50,000 lb for a tandem bearing assembly. Radial load is applied normal to the test bearing axis and is controllable in increments of 1000 lb (± 200), from no load to 20,000 lb per bearing. The load application rate is variable and controllable from 0 to full load in 0.5 sec to 30 sec. Axial and radial loads can be applied separately or simultaneously. The radial load should not exceed 50% of the axial load when both loads are applied simultaneously. Load applicators are pneumatic, requiring either gaseous nitrogen or gaseous helium. The working design pressure of the coolant inlet system is 400 psig.

The controls and instruments are suitable for remote operation. Instrumentation is provided to record torque, vibration, temperature at bearing outer race and bearing cavity, pressure of bearing cavity, phase, strain gage for outer race, cage speed, and thrust. Figures 6 and 7 show typical instrumentation installations.

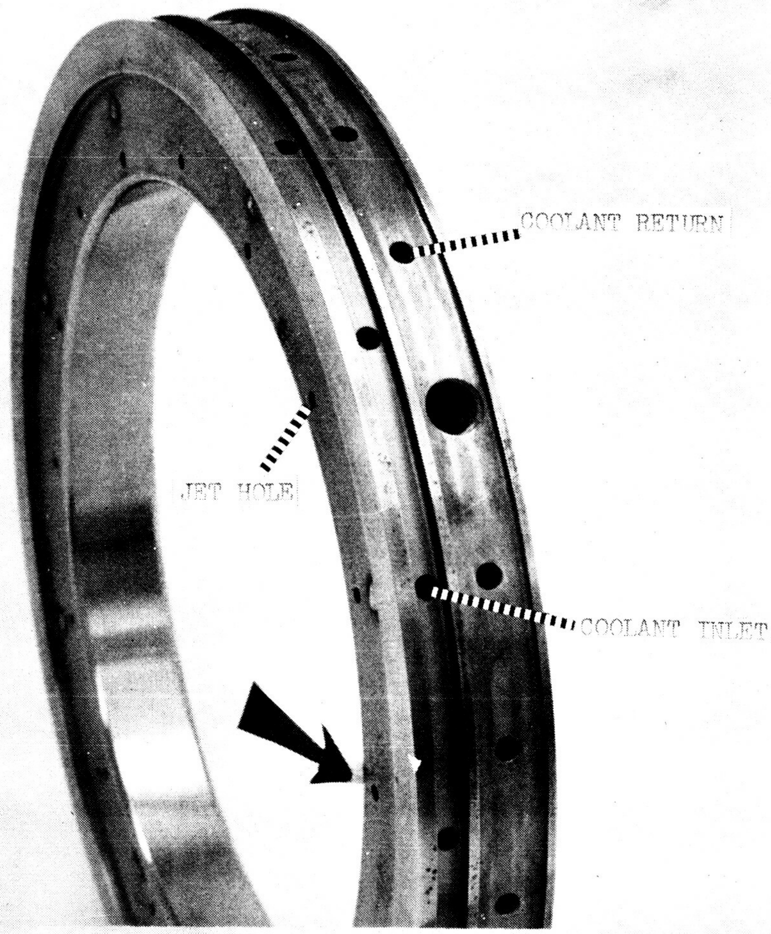

Figure 2

Coolant Spray Ring

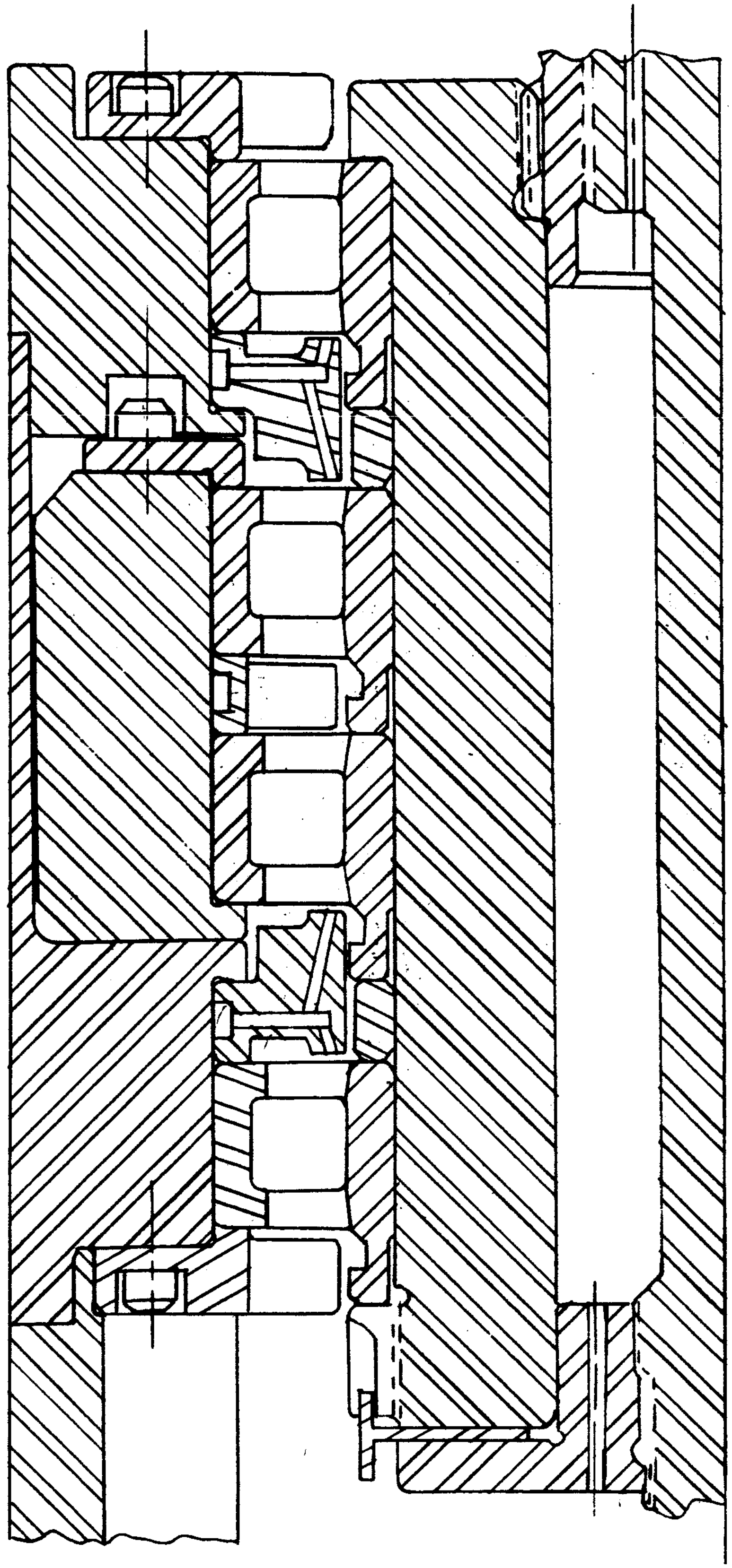

Figure 3

Roller Bearing Test Head

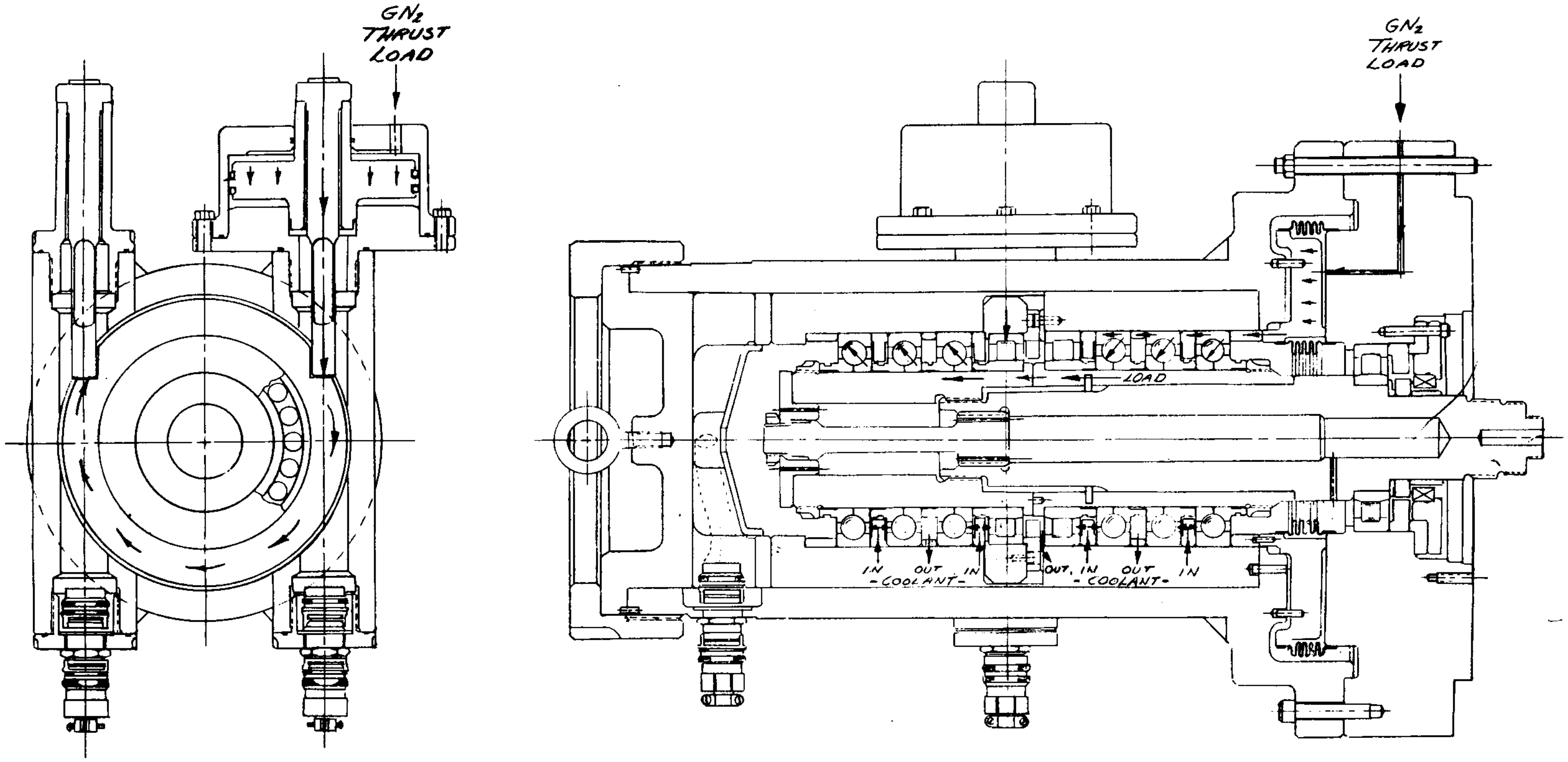

Figure 4
Test Head, Single Thrust Bearing

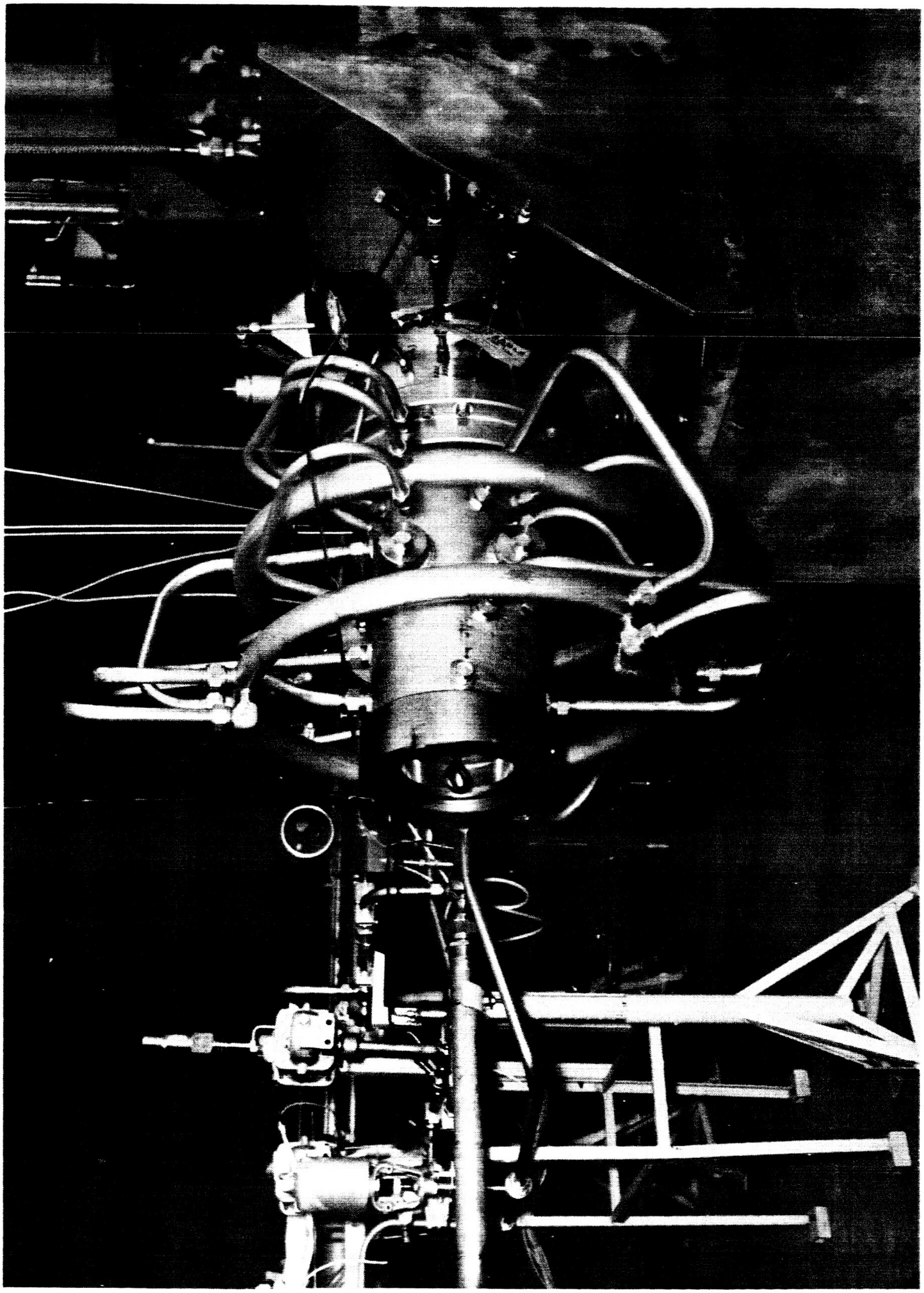

Figure 5

Test Stand, Motor-Driven Tester

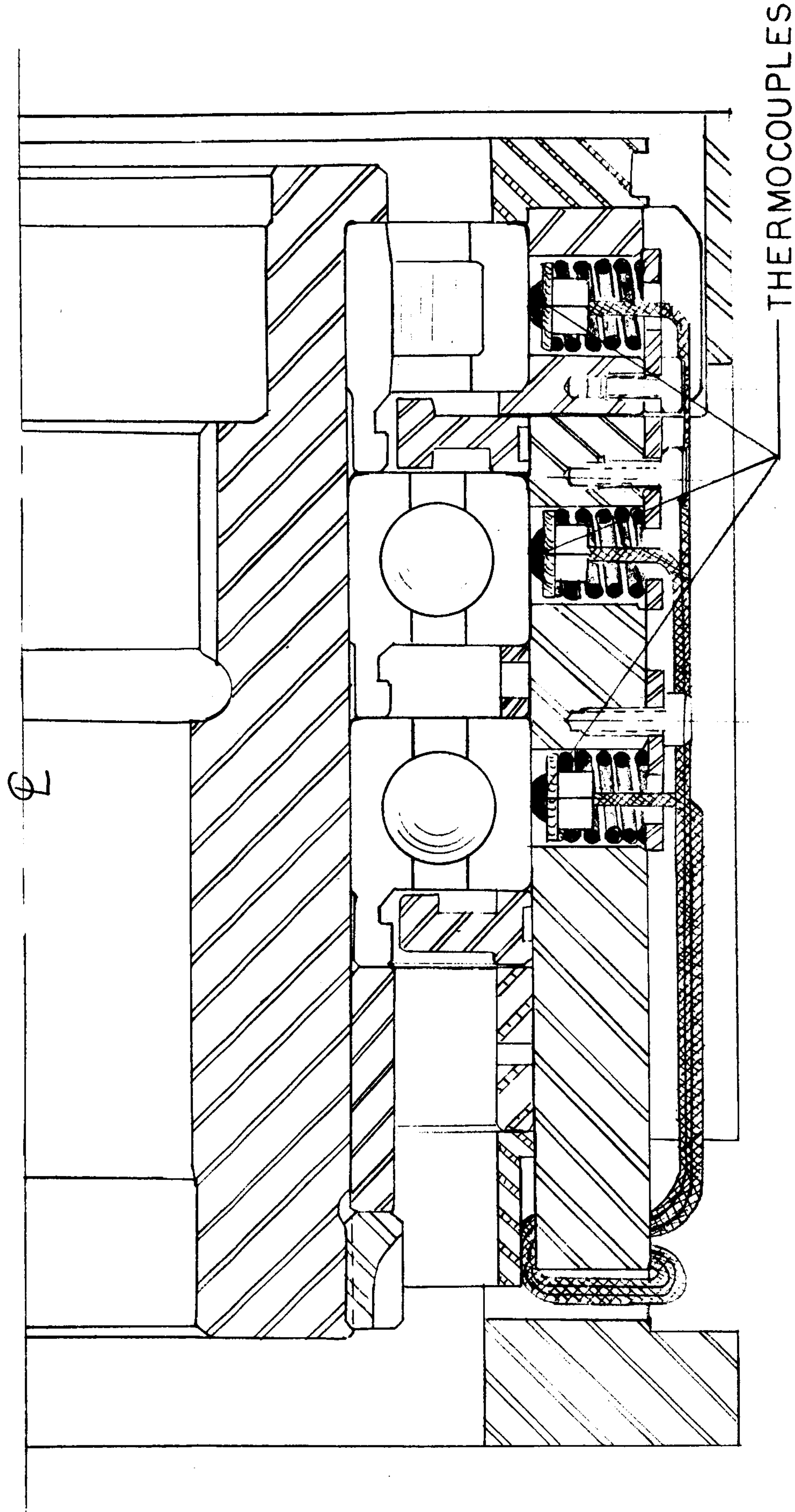

Figure 6

Typical Thermocouple Location

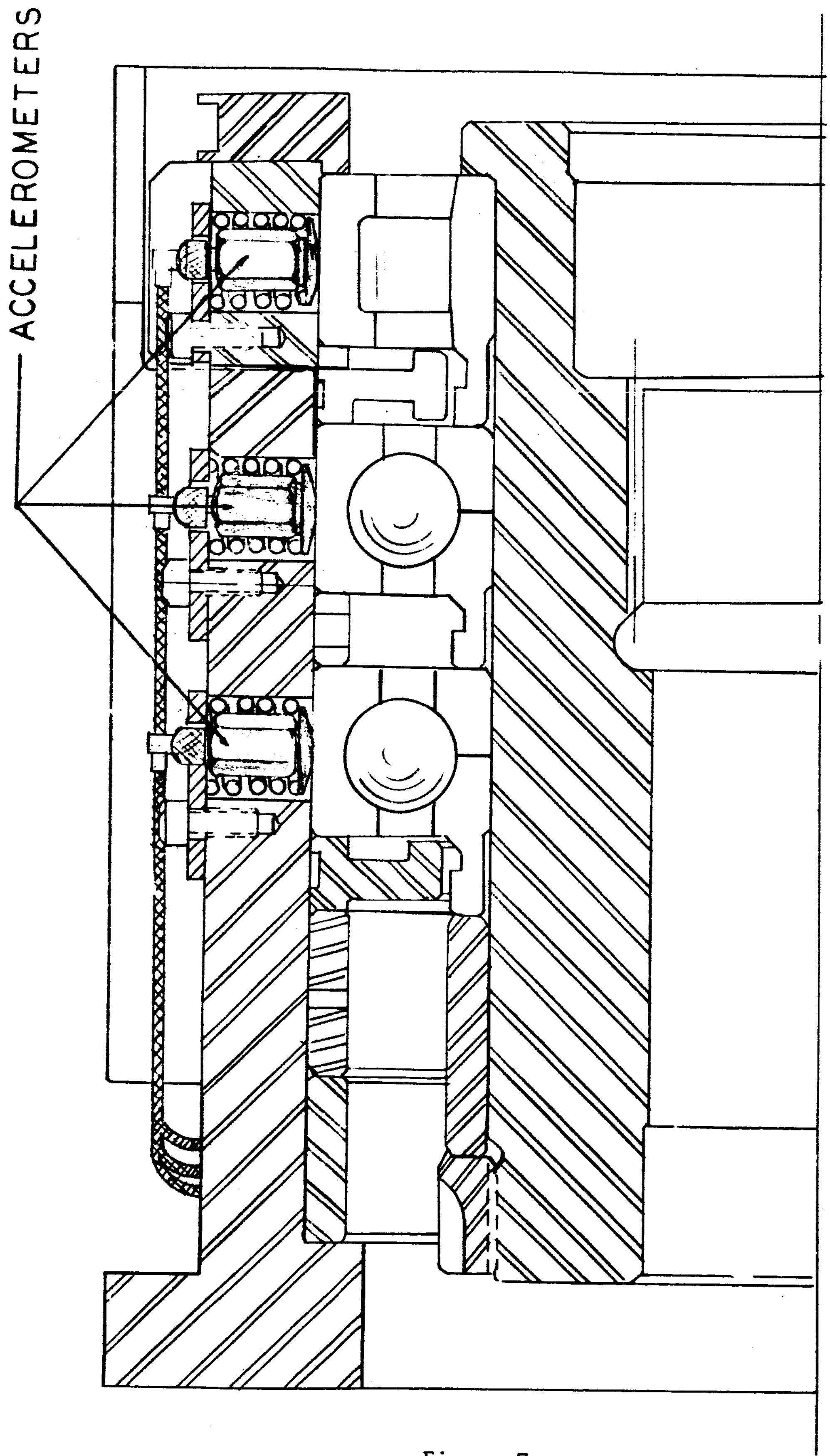

Figure 7

Typical Accelerometer Location

The test cartridge is suitable for testing 105 x 160 mm, 110 x 150 mm, 110 x 170 mm, and 120 x 180 mm bearings.

2. Turbine-Driven Tester (P/N 284994)

The turbine-driven single bearing tester, shown schematically in Figure 8 and installed into the test bay in Figure 9, is designed to test two single ball thrust bearings, arranged in opposite load directions, at total loads up to 25,000 lb and shaft speeds up to 20,000 rpm. The bearings are mounted on a direct-drive shaft encased in an axially-floating housing incorporating an internal hydraulic load actuation system. The load is applied directly to the outboard bearing through the hydraulic actuation system and appears at the inboard bearing as a resistive reaction load. Pressure drop through the coolant scavenge line results in additional load on the inboard bearing because the coolant back pressure acts upon the floating test housing. The drive unit is a modified Titan I turbine, using gaseous nitrogen as the drive gas.

Speed is controlled by manually operating the turbine inlet pressure and is based upon a digital readout tachometer. Transient load studies are not practical because of the slow response time. Acceleration is controlled by the pre-set turbine inlet pressure upstream of a fast actuating valve. The pre-set pressure levels are determined on a trial and error basis. Deceleration is accomplished by manual remote closure of the turbine gas flow control valve. Automatic overspeed protection vents the turbine drive gas to atmosphere in the event of an overspeed signal.

3. Turbine-Driven Tester (P/N 280116)

The tester, shown schematically in Figure 10 and installed in the test bay in Figure 11, is used in the roller bearing acceleration tests. This tester is designed to test one roller bearing and one ball bearing (slave) at partial load, with acceleration rates up to 25,000 rpm/sec. The bearings are mounted on a direct-drive shaft encased in a one-piece housing. Both axial and radial loads are applied through the external hydraulic load actuators. The drive unit is a modified Titan I turbine using gaseous nitrogen. Speed control and overspeed protection are the same as for the tester previously discussed.

C. TEST PROCEDURE

1. Motor-Driven Tester

The pre-test chilldown and test procedure for the motor-driven tester was generally as delineated below.

a. Vent the seal cavity to discharge and purge the tester with gaseous nitrogen at 50 psig at 125°F for 30 minutes.

b. Apply gaseous hydrogen purge at 50 psig at ambient temperature for 15 minutes prior to liquid hydrogen bleed-in.

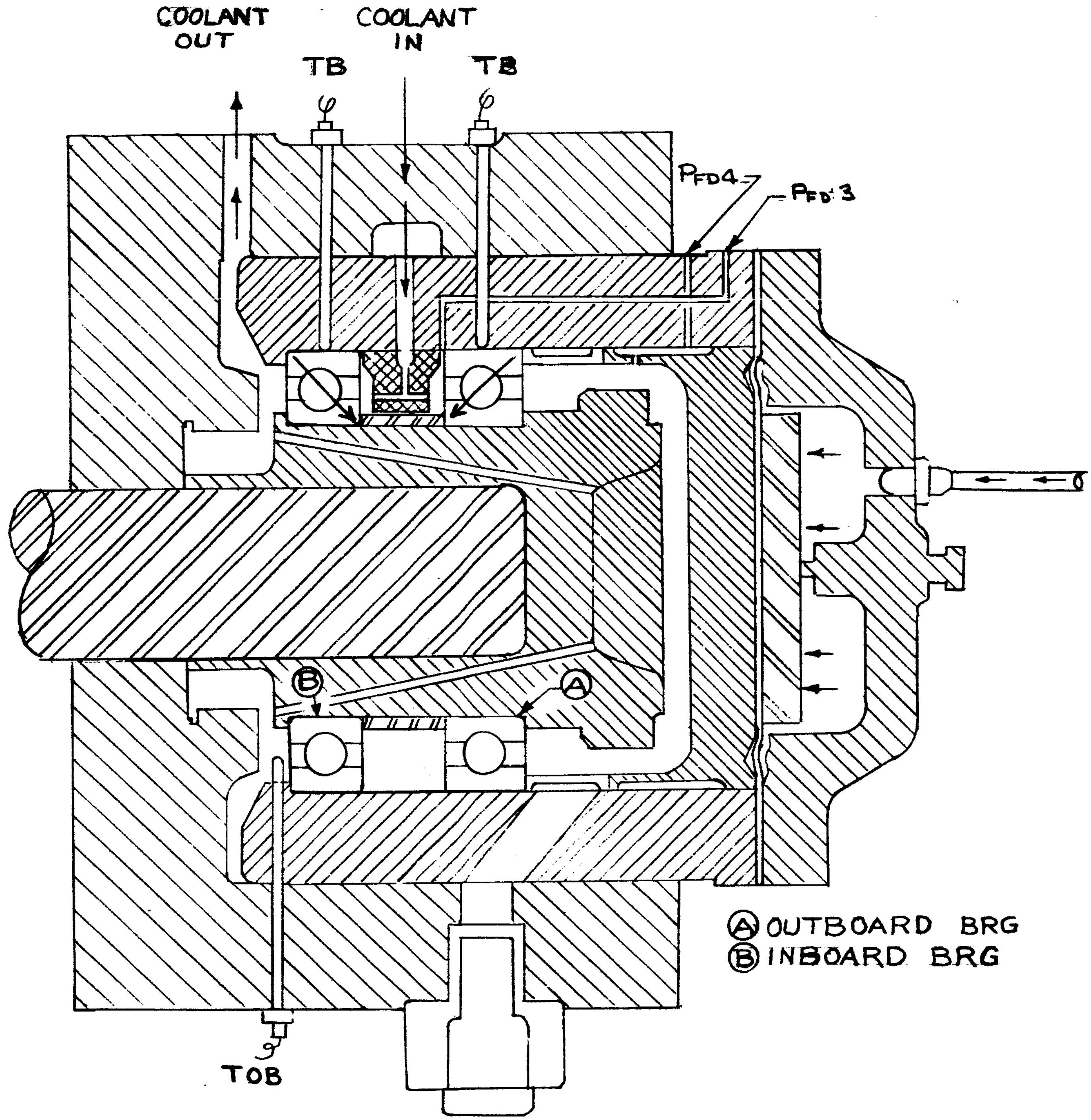

Figure 8

Test Head, Single Thrust Bearing

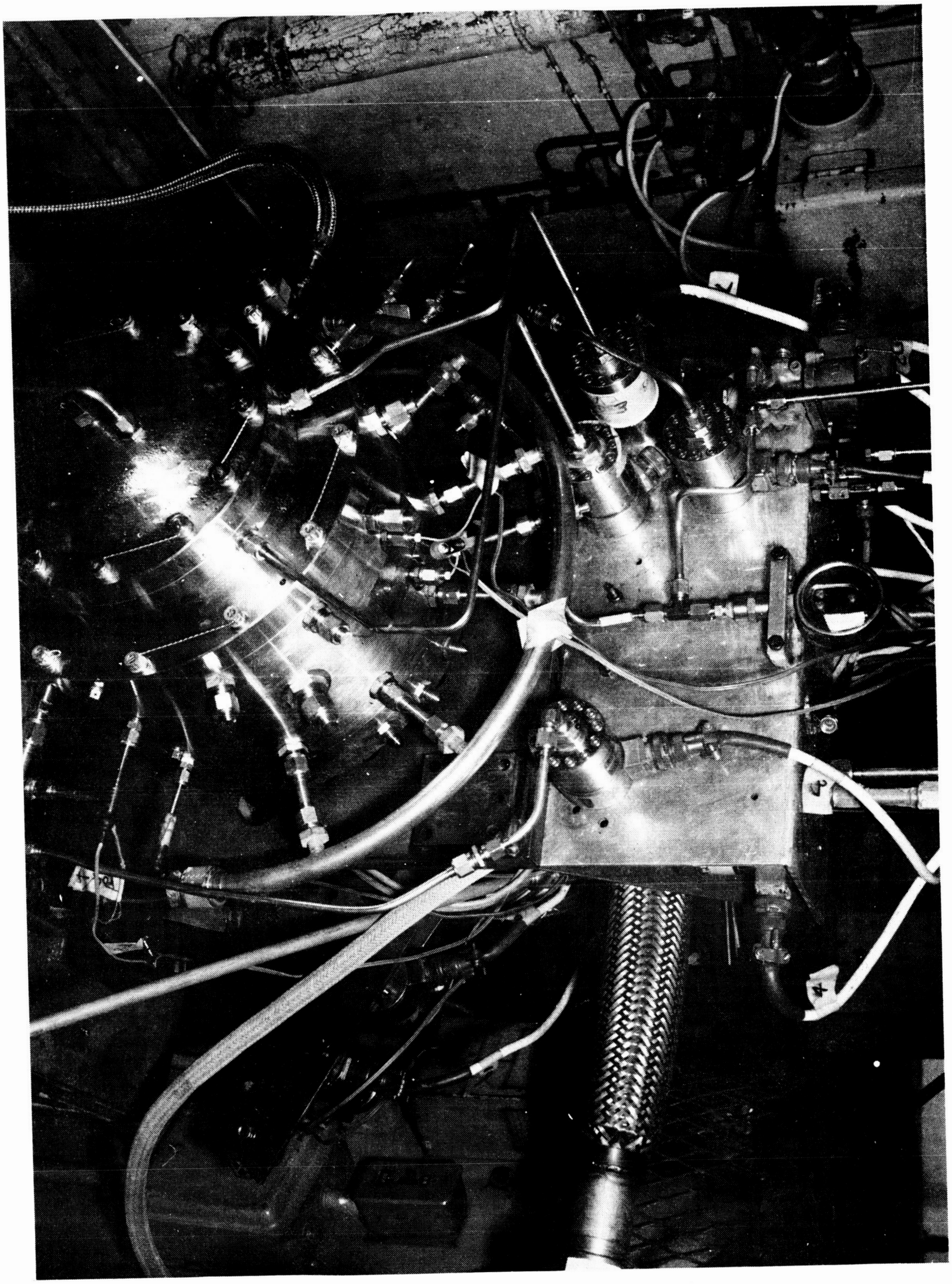

Figure 9

Single Bearing Test Head

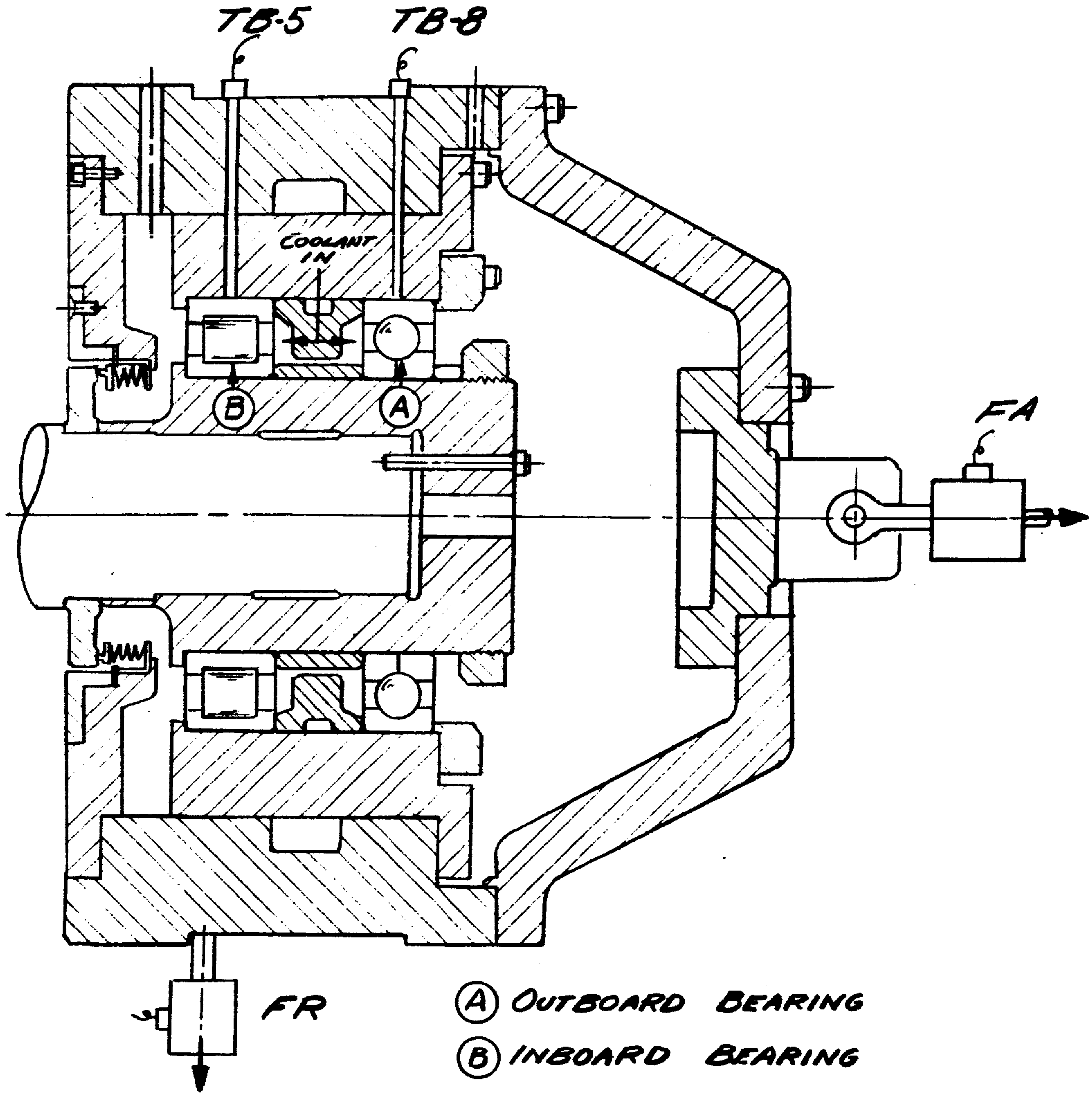

Figure 10

Test Head, Acceleration Tester

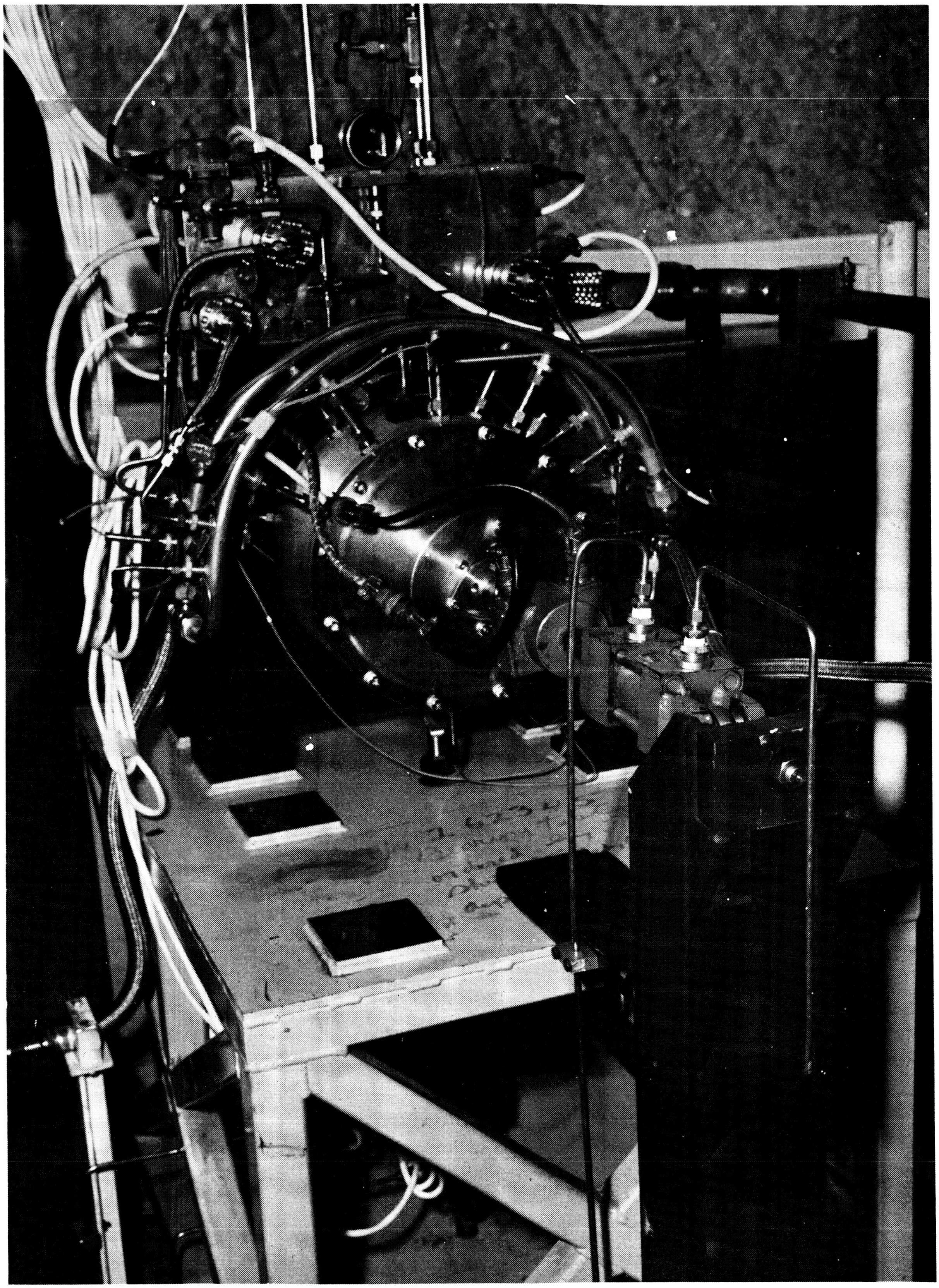

Figure 11

Test Stand, Acceleration Tester

c. Bleed-in liquid hydrogen.

d. When the bearing temperature is -400°F, set liquid hydrogen discharge pressure.

e. Set the flow rate of liquid hydrogen.

f. Set gaseous helium pressure to 60 psig (radial load).

g. Start motor-drive unit and attain the required speed.

h. Set gaseous helium pressure (radial load) to the desired value and maintain it for the duration of the test.

i. Adjust the liquid hydrogen flow control.

j. Maintain hydrogen flow for the test duration.

k. Decrease speed to 0 rpm at 10% tank level.

l. Shutdown liquid hydrogen system when speed reaches 0 rpm.

m. Reduce radial load pressure to 60 psig.

n. Repeat items d through m to obtain the desired test duration.

2. Turbine-Driven Tester

The pre-test childown and test procedure for the turbine-driven tester was generally as delineated below.

a. Purge the bearing tester with gaseous nitrogen at 50 psig at a minimum temperature of 125°F for 30 minutes.

b. Purge with ambient gaseous hydrogen at 50 psig for 15 minutes prior to the liquid hydrogen bleed-in.

c. Purge the seal cavity with gaseous nitrogen for the duration of the test.

d. Bleed-in liquid hydrogen.

e. Set the radial load pressure to the desired pressure.

f. When the bearing temperature reaches -400°F, set the liquid hydrogen discharge pressure to the desired pressure and continue childown for five minutes.

g. Close the flowmeter by-pass valve and verify the flowmeter rotation.

h. Replenish the run tank and pressurize the tester to the required pressure.

i. Adjust the liquid hydrogen flow.

j. Start the turbine-drive unit and accelerate to 10,000 rpm in 0.5 sec.

k. Decrease speed to 0 rpm.

l. Repeat items j and k to obtain the desired number of acceleration tests.

IV. BEARING DEVELOPMENT PROGRAM

A. CONFIGURATION

The configuration of the bearings tested under the program conforms to the following dimensions.

1. Roller Bearing - Pump End - 110 mm (P/N 288260)

Inside Diameter	110 mm
Outside Diameter	150 mm
Width	20 mm
Roller Diameter (in.)	0.433
Roller Diameter Variation (in.)	± 0.00001
Number of Rollers	24
Roller Length (in.)	0.433
Race and Roller Material	440C
Cage Material	Armalon
ABEC Class	7
Vendor	Industrial Tectonics
Diametral Clearance (as built) (in.)	0.003/0.0035
Roller Pocket Dia. Clearance (in.)	0.021
Roller Pocket End Clearance (in.)	0.028
Guiding Shoulder Clearance (in.)	0.002
Eccentricity - Inner Race (in.)	0.0002
Outer Race (in.)	0.0003
Parallelism of Sides -	
Inner Race (in.)	0.00015
Outer Race (in.)	0.0002

2. Roller Bearing - Turbine End - 120 mm (P/N 288340)

Inside Diameter	120 mm
Outside Diameter	180 mm
Width	28 mm
Roller Diameter (in.)	0.526

Roller Diameter Variation (in.)	± 0.00001
Number of Rollers	26
Roller Length (in.)	0.645
Race and Roller Material	440C
Cage Material	Armalon
ABEC Class	7
Vendor	Bower
Diametral Clearance (as built) (in.)	0.0031/0.0035
Roller Pocket Dia. Clearance (in.)	0.021
Roller Pocket End Clearance (in.)	0.028
Guiding Shoulder Clearance (in.)	0.002
Eccentricity - Inner Race (in.)	0.0002
Outer Race (in.)	0.0003
Parallelism 4 Sides -	
Inner Race (in.)	0.00015
Outer Race (in.)	0.0002

3. <u>Ball Bearing - 110 mm (P/N 288410)</u>

Inside Diameter	110 mm
Outside Diameter	170 mm
Width	28 mm
Ball Diameter (in.)	23/32
Roller Diameter Variation (in.)	± 0.00001
Number of Balls	20
Race and Ball Material	440C
Cage Material	Armalon
ABEC Class	7
Vendor	Industrial Tectonics
Contact Angle (degrees)	30
Race Curvature - Inner (%)	53
Outer (%)	52
Diametral Clearance (as built) (in.)	0.0068/0.0074
Axial Play (as built) (in.)	0.014/0.020
Dynamic Contact Angle -	
Inner (degrees)	35.8
Outer (degrees)	32.2
Ball Pocket Dia. Clearance (in.)	0.029
Eccentricity - Inner (in.)	0.0002
Outer (in.)	0.0003
Parallelism of Sides -	
Inner Race (in.)	0.00015
Outer Race (in.)	0.0002

B. OPERATING STRESSES

1. Thrust Bearings

An analysis was made to determine the contact, hoop, and radial stresses at ambient and operating temperatures for the prototype (P/N 299410) tandem bearing stack; the total shared load was assumed to be 35,000 lbs. The study was made for the highest loaded bearing of the triple set assuming 40%30%27% load sharing. This analysis assumed maximum (tight) fit conditions. Analysis results are shown in Table I, at 13,300 rpm and 15,000 lbs (43% load). The dynamic contact angles of 35.8 degrees on the inner and 32.2 degrees on the outer race result in the contact ellipse being confined within the design shoulder height.

2. Roller Bearings

Hoop and compressive stresses were determined for pump and turbine radial bearings at room temperature with no load and operating temperature at design load. Maximum interference fits were used in these calculations; the results are presented in Table II.

C. TEST RESULTS AND DISCUSSION

The test program was divided into three phases: evaluation of turbine-end (120 mm) roller bearings, evaluation of 110 mm thrust bearings, and evaluation of pump-end (110 mm) roller bearings. Evaluation of these bearings was conducted under simulated turbopump operating conditions of speed, load, acceleration rate, and coolant flow and temperature. A complete tabulation of bearing configuration, test conditions, and the results of each test phase is given in Appendix A. A brief summary table and a discussion of the results of each test phase and category are given below.

1. Phase I - Evaluation of Turbine-End (120 mm) Radial Bearings

a. Category A - Single Radial Bearings, Constant Speed

(1) Summary

Total tests (four bearing tests/run)	108
Total bearings tested	8
Total duration accrued (sec)	26,952
Failures	0
Total duration on one bearing (sec)	5880
Total starts on same bearing	23
Typical load on same bearing (lb)	15,500
Maximum load per bearing (lb)	15,500
Duration at maximum load (sec)	5278

(2) Discussion

Post-test inspection of the Buildup No. 1 bearings which were tested in the motor-driven tester shown in Figures 3 and 5, revealed that the inboard bearings were in excellent condition with no visible signs of distress. The rollers of the outboard bearings showed severe end wear and the races showed signs of skidding and chattering as well as slight burning in the cage pockets (see Figures 12 and 13). This wear was the result of unequal loading attributed to tester shaft deflection, which is considered a worse condition than what will be experienced in the turbopump.

TABLE I

CALCULATED STRESSES FOR M-1 LH_2 THRUST BEARING

Temp (°F)	Single Row Load (lbs)	Speed (RPM)	Free Contact Angle (Degrees)	Mounted Contact Angle (Degrees)	Dynamic Contact Angle-(Degrees)		Radial Interference (Inches)		Maximum Compress. Stress (Hertz) (psi)	Tangential Stress (Hoop) (psi)		Radial Stress (psi)		Clamping Load (lbs)	
					Inner Race	Outer Race	Inner Race	Outer Race		Inner Race	Outer Race	Inner Race	Outer Race	Inner Race	Outer Race
68	0	0	30	27.3	--	--	0.0021	Loose	-	13100	-	*2240	-	9600	14800
-420	0	0	30	29.2	--	--	0.0006	0.00085	-	4500	*4769	*825	*355	135000	115000
-420	15000	13300	30	25.4	35.8	32.2	0.0006	0.00085	391600	4500	*4769	*825	*355	135000	115000

*Compressive Stress

TABLE II

RESULTS OF MAXIMUM INTERFERENCE FITS

	PUMP ROLLER TEMPERATURE		TURBINE ROLLER TEMPERATURE	
	AMBIENT	-420°F	AMBIENT	-420°F
RADIAL LOAD, LB	0	4,600	0	10,800
INNER RACE HOOP STRESS, PSI	25,000	8,250	5,130	2,980
INNER RACE RADIAL STRESS, PSI	-2,250	-740	-570	-330
OUTER RACE TANGENTIAL STRESS, PSI	0	5,350	7,750	4,790
OUTER RACE RADIAL STRESS, PSI	0	-285	-710	-438
INNER RACE CONTRACT STRESS, PSI	141,000	208,000	0	215,000
OUTER RACE CONTACT STRESS, PSI	130,000	196,000	0	201,000

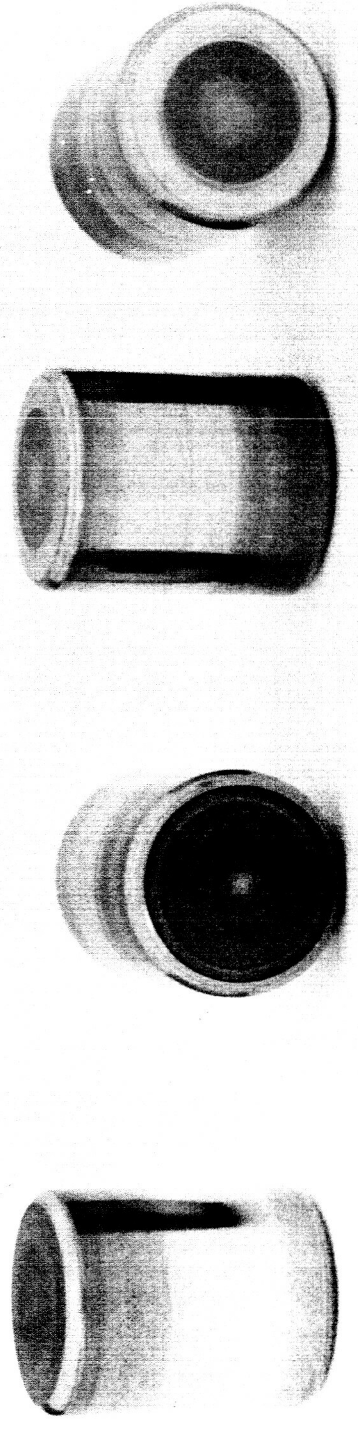

Figure 12

120 mm Roller Bearing Rollers

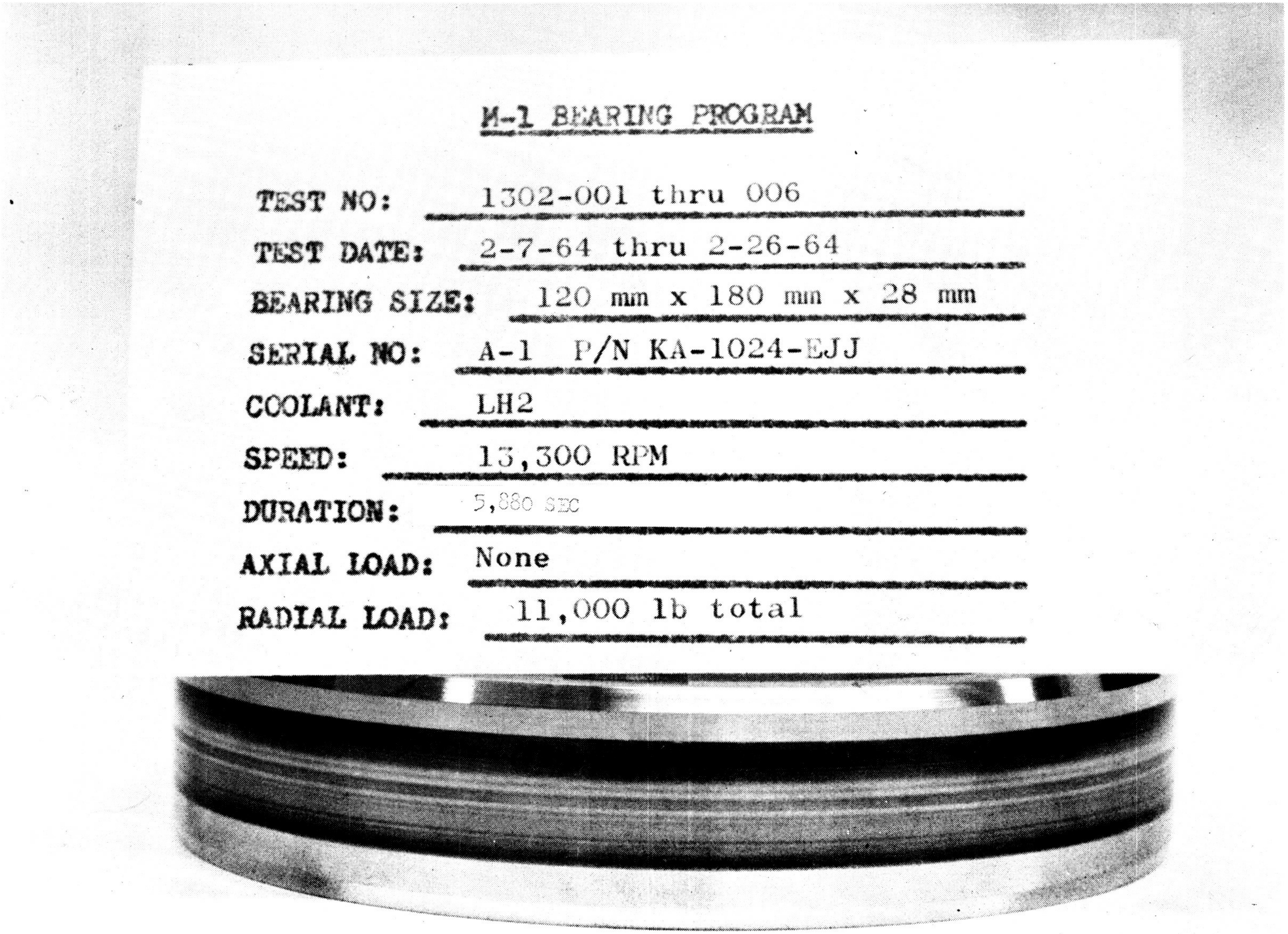

Figure 13

120 mm Roller Bearing Inner Race

Post-test analysis of the Buildup No. 2 bearings revealed that the applied load was not fully transferred to the bearings, which caused a probable reduction in the load of 1500 lb per bearing, leaving 15,500 lb per bearing as the estimated radial load. All bearings had inner and outer race wear, roller-end wear, cage pocket wear, asperity welding, chipping of the guiding shoulder of the outer race, and roller skidding (see Figures 14, 15, and 16). However, the four bearings operated at rated speed and 40% overload for 5880 sec with a coolant flow one-fourth of the 100 gpm predicted as the probable turbopump requirement.

b. Category B - Single Radial Bearings, Acceleration Evaluation

(1) Summary

Total tests	22
Total bearings tested	1
Total duration accrued (sec)	3
Total failures	0
Total starts on one bearing	22
Typical acceleration on same bearing (rpm/sec)	25,000
Typical load on same bearing (lb)	500
Maximum acceleration (rpm/sec)	28,000

(2) Discussion

These tests were conducted using the turbine-driven tester shown in Figures 10 and 11. Post-test examination of the bearing revealed slight signs of roller-end wear. Otherwise, the bearing was in excellent condition. Based upon these 22 acceleration tests, it was concluded that this roller bearing is capable of withstanding the turbopump acceleration conditions without failure.

c. Category C - Single Radial Bearings, Initial Turbopump Conditions, (Low Speed, Low Coolant Flow, and Low Pressure at Start)

(1) Summary

Total tests	3
Total bearings tested	1
Total duration accrued (sec)	1000

Figure 14

120 mm Roller Bearing Outer Race

Figure 15

120 mm Roller Bearing Rollers

Figure 16

120 mm Roller Bearing Cage

Failures	0
Total duration on one bearing (sec)	1000
Total starts on same bearing	3
Typical load on same bearing (lb)	3000
Maximum load per bearing (lb)	4500
Duration at maximum load (sec)	162

(2) Discussion

The turbopump test program initially required low speed testing which was gradually increased during subsequent tests, until full speed was reached. Low turbopump speed results in low pump discharge pressure and consequently, low coolant pressure and flows. The tests with Buildup No. 6 were conducted at speeds of 3300, 5000, and 9900 rpm. Radial load coolant flow and bearing cavity pressure were low at the start, and increased with speed. Post-test inspection of the bearing revealed slight roller-end wear and minor cage pocket burning. There were a few small particles of 440C material imbedded in the cage; this material originated from the outer race shoulder and roller ends. Both the inner and outer races showed a non-uniform wear path indicating a cocked roller condition. This will not be present in the turbopump; however, the success of this test indicates that the bearing has the capability to function in the event of misalignment of the rotor in the turbopump assembly. These tests were conducted using the turbine-driven tester shown in Figures 10 and 11.

2. Phase II - Evaluation of 110 mm Thrust Bearings

a. Category A - Single Thrust Bearing, Constant Speed

(1) Summary

Total tests	16
Total bearings tested	3
Total duration accrued (sec)	2412
Failures	0
Total duration on one bearing (sec)	384
Total starts on same bearing	4
Typical load on same bearing (lb)	25,000
Maximum load per bearing (lb)	26,000
Duration at maximum load (sec)	156

(2) Discussion

Initial thrust bearing testing was conducted using the turbine-driven tester shown in Figures 8 and 9. Single bearings identical to those used for the triple bearing tandem set were utilized. One was used as the test bearing and the lighter loaded bearing for a slave or loading bearing. Both are reported as individual tests.

Test measurements taken during tests with Buildup No. 1 and 2 indicated no problems. Accordingly, the acceleration evaluation that followed was initiated without prior bearing inspection. Evaluation subsequent to tests in that category showed the bearings to be in excellent condition.

b. Category B - Single Thrust Bearing, Acceleration Evaluation

(1) Summary

Total tests	42
Total bearings tested	2
Total duration accrued (sec)	100
Failures	0
Total starts on one bearing	21
Typical acceleration on same bearing (rpm/sec)	6000
Typical load on same bearing (lb)	25,000
Maximum acceleration (rpm/sec)	15,000

(2) Discussion

These tests were conducted using the turbine-driven tester shown in Figures 10 and 11. Post-test examination of the Buildup No. 2 showed no evidence of damage during either the preceding constant speed test or these acceleration tests, with the exception of blistering of the copper plating on the wear path. This damage is not applicable to the turbopump operation because there are no plans to use plating.

c. Category C - Tandem Thrust Bearings, Constant Speed

(1) Summary

Total tests (two bearing tests/run)	26
Total bearings tested (tandem sets)	4
Total duration accrued (sec)	2778

Failures	2
Total duration on one bearing (sec)	2766
Total starts on same bearing	12
Typical load on same bearing (lb)	36,000
Maximum load per tandem set (lb)	40,000
Duration at maximum load (sec)	1666

(2) Discussion

Testing of Buildup No. 3 was accomplished with the motor-driven tester shown in Figures 4 and 5 and was terminated after 1890 sec because of a bearing temperature rise of 200°F. The outboard ball bearing set was in excellent condition with load sharing estimated at 37%, 23%, and 40%. Two ball bearings in the inboard triple set had experienced excessive heat and failed. The inner race of one bearing had broken (see Figure 17) and the balls were covered with deposits of metal (see Figure 18). The inner race of the second bearing in the failed set had hair-line cracks in the outer race and the balls were pitted.

The failure of this bearing set was attributed to inadvertent low coolant flow caused by icing of the coolant supply passages. This icing appeared to be the result of either the residual moisture or the nitrogen purge gas freezing when chilled with liquid hydrogen. Similar behavior was observed during filter flow tests under similar conditions. Because of the 1300-sec duration, at or exceeding turbopump design conditions, it was concluded that the triple ball bearing set was satisfactory for use in the turbopump. Future tests would be concerned with confirming this conclusion as well as investigating bearing capacities at the low speeds and coolant conditions to be experienced during preliminary turbopump testing.

Testing of Buildup No. 4 was terminated after 6 sec because the shaft torque increased to 790 in.-lb and the rotating assembly froze. However, the temperature rise of the bearings indicated only a 6°F rise. Post-test examination of the assembly revealed that on one bearing, the inner diameter of the coolant spray ring rubbed on the puller groove inner race extension of the bearing. This caused the spray ring to crack from the heat and the inner race to break at four places (see Figures 19, 20, and 21).

Analysis of the assembly dimensions showed that the 0.030/0.040-in. clearance between the outer race diameter and the housing permitted the coolant spray ring to be assembled 0.030/0.040-in. eccentric to the bearing, which causes bearing-to-spray-ring misalignment. Because the clearance between the inner race and spray ring is only 0.015-in., rubbing occurred. This eccentric condition cannot occur in the turbopump assembly. For subsequent tests, shims were added to the bearing outside diameter to prevent reoccurrence of misalignment.

Figure 17

Broken Inner Race

Figure 18

Bearing Balls

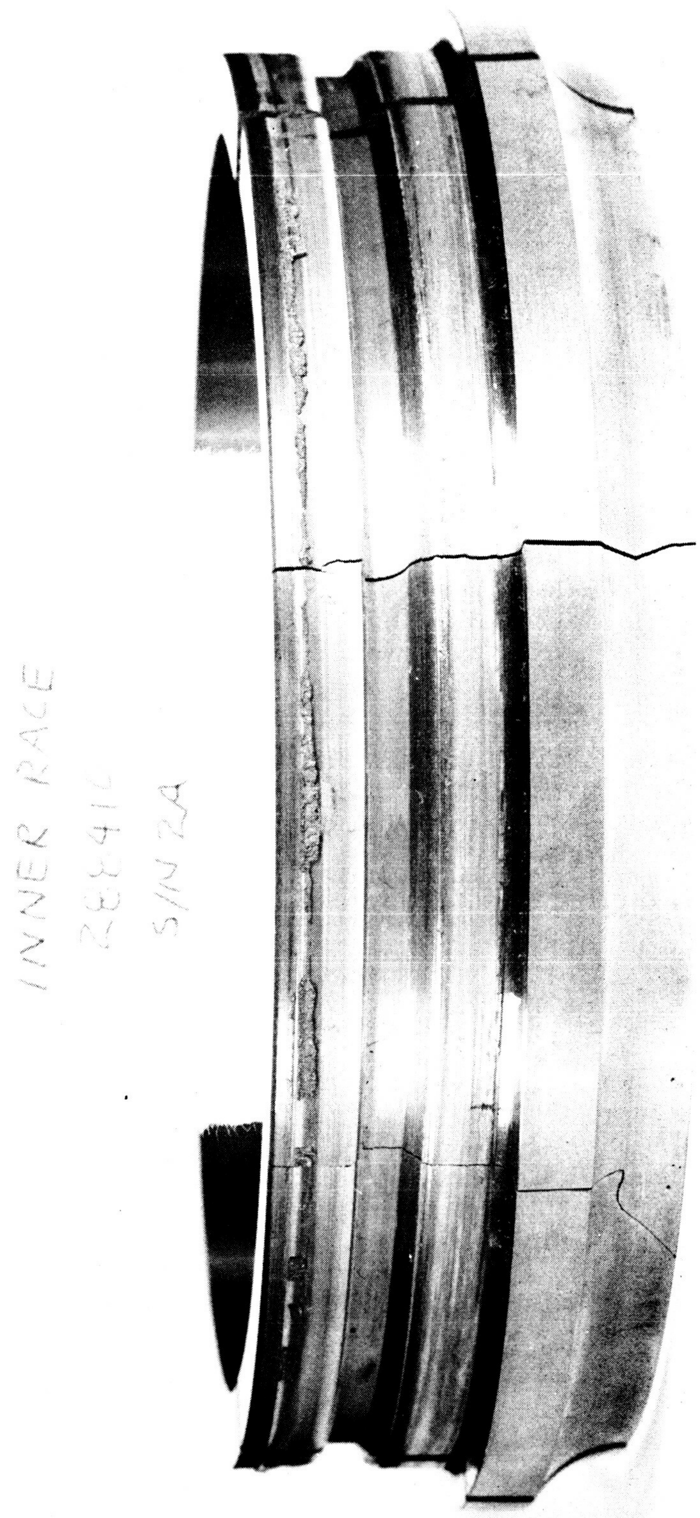

Figure 19

Broken Inner Race

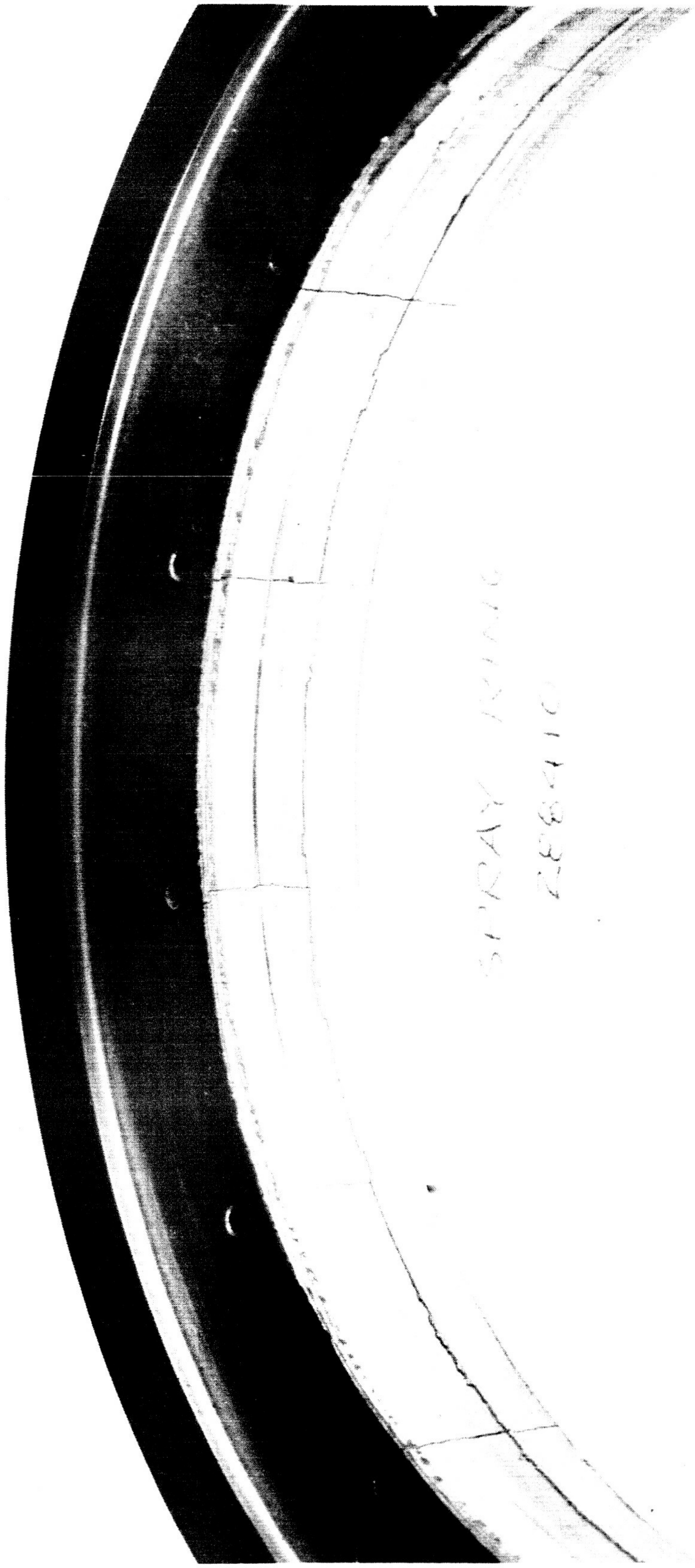

Figure 20

Cracked Coolant Ring

Figure 21

Broken Inner Race

d. Category D - Tandem Thrust Bearings - Initial Turbopump Conditions (Low Speed, Low Coolant Flow, and Pressure at Start)

(1) Summary

Total tests (two bearing tests/run)	18
Total bearings tested (tandem sets)	3
Total duration accrued (sec)	2323
Failures	1
Total duration on one bearing (sec)	2323
Total starts on same bearing	8
Typical load on same bearing (lb)	30,000
Maximum load per tandem set (lb)	30,000
Duration at maximum load (sec)	492

(2) Discussion

Tests with Buildup No. 5, 6, and 7 were conducted using the motor-driven tester (see Figures 4 and 5) under conditions of speed, coolant pressure, and flow as predicted for the initial turbopump testing. Data from Buildup No. 5 testing was acceptable but testing of Buildup No. 6 was terminated after 30 sec during the third start when the bearing temperature rose to -300°F during a planned test at 9900 rpm and 30,000 lb thrust load. Inspection of the bearings revealed that the damage was confined to one bearing of one tandem set. The inner race had one complete radial break and 35 radial cracks. The break and cracks were caused by overheating resulting from a decrease in the coolant flow rate to negligible quantity because of a flow control system malfunction during the test. The second set and the two other rows of the failed set showed nominal wear and were acceptable for additional testing. Bearing performance was acceptable at simulated turbopump flow rates.

The accumulated testing time for Buildup No. 7 was 660 sec using one bearing set from the previous buildup and one new set. Post-test inspection of the bearings revealed normal wear on one bearing. The other raceways were heavily pitted, indicating high loading. There was no debris present in the cage or other parts. The bearing appeared adequate for short duration, whereas the amount of damage was considered excessive for long duration, high load operation.

It was concluded from the results of testing with Buildup No. 3, 4, 5, 6, and 7 that tandem thrust bearings are adequate for operation under the turbopump design load and speed. Where bearing failures occurred, the cause was misalignment in the tester or accidental low coolant flow. Tester misalignment was caused by conditions that are not part of the turbopump design. Low coolant flow, caused by icing of the coolant passages, can be avoided by adequate purging. Flow control system malfunctions of this type will not occur during turbopump testing.

e. Category E - Single Thrust Bearing - Constant Speed, Reverse Load

(1) Summary

Total tests	4
Total bearings tested	3
Total duration accrued (sec)	528
Failures	0
Total duration on one bearing (sec)	528
Total starts on same bearing	4
Typical load on same bearing (lb)	9500
Minimum load per bearing (lb)	0
Duration at minimum load (sec)	234

(2) Discussion

Testing of Buildup No. 8 and 9 was conducted using the motor-driven tester (see Figures 4 and 5) with single ball bearings from previously used tandem bearing sets that had been run at high loads and speeds.

The objective of this testing was to operate single bearings under conditions simulating the load expected during 6000 rpm turbopump testing, in which the net thrust load is predicted in the direction opposite to the load sharing direction. Therefore, this reversed thrust load could possibly be carried by one ball bearing. Because of the low speed, the coolant pressure was estimated to be 90 psig. Buildup No. 1 and 2 testing had been successful with single ball bearing at high load, but the coolant pressure was 310 psig (relatively high in relationship to that available at low speed).

The first tests were run for 294 sec, of which 60 sec was at 15,000-lb thrust. The temperature of one bearing rose to -230°F in 35 sec. Inspection of the bearing revealed that the balls and races were discolored because of heat. This bearing had accumulated 1380 sec during previous tests. Another bearing was substituted and the test continued for 234 sec at zero axial thrust load. Post-test inspection of this bearing revealed normal wear. The wear path for no axial load could not be determined because these bearings had been used for previous tests at various loads and speeds.

It was concluded from these series of tests that the ball bearings are capable of withstanding conditions imposed by the planned low speed turbopump tests, under the reverse-loaded condition.

3. Phase III - Evaluation of Pump End (110 mm) Radial Bearings

a. Category A - Single Radial Bearing, Constant Speed

(1) Summary

Total tests (two bearing tests/run)	26
Total bearings tested	4
Total duration accrued (sec)	2790
Failures	0
Total duration on one bearing (sec)	2770
Total starts on same bearing	12
Typical load on same bearing (lb)	5000
Maximum load per bearing (lb)	8000
Duration at maximum load (sec)	20

(2) Discussion

Post-test examination revealed that Buildup No. 1 bearings were in excellent condition. This was the same test series in which thrust bearing failure occurred because ice blocked the coolant passages. The low coolant flowrate had no adverse effect upon radial bearing performance. (Radial bearing tests were run in conjunction with tandem thrust bearing tests but are reported separately for clarity. Buildup No. 1 through 7 correspond to tandem thrust bearing Buildup No. 3 through 9, phase II, Category C. D. and E.)

b. Category B - Single Radial Bearings, Initial Turbopump Conditions (Low Speed, Low Coolant Flow, and Low Pressure at Start)

(1) Summary

Total tests (two bearing tests/run)	26
Total bearings tested	2

Total duration accrued (sec)	3496
Failures	0
Total duration on one bearing (sec)	1748
Total starts on same bearing	13
Typical load on same bearing (lb)	1000
Maximum load per bearing (lb)	1000
Duration at maximum load (sec)	1748

(2) Discussion

The condition of the bearings upon disassembly of Buildup No. 2 through 7 was excellent and in each case, the bearings were reassembled for additional testing.

V. CONCLUSIONS AND RECOMMENDATIONS

It is indicated from the results of the bearing development program that the bearing capacities exceed the design requirements of the fuel turbopump. However, operational engine tests will require extended duration turbopump tests. While longer tests are more severe, the bearing development program has shown that the current design loads can be carried for a longer period. The roller bearings showed a higher overload capacity than the ball bearings. However, accurate thrust balancing in the turbopump will result in lower thrust loads and will also contribute to prolonged thrust bearing life. Those failures that did occur, which were discussed in this report, are attributed to operational or mechanical problems (i.e., inadvertent loss of coolant flow, shaft deflection, and misalignments) that are unique to the testers and are not expected to occur in the turbopump.

The coolant flow requirements were considerably less than predicted. For example, the 120 mm roller bearing operated successfully at rated speed and 40% overload and only 40°F temperature rise with 26 gpm liquid hydrogen flow. This is one-quarter of the predicted requirement of 100 gpm for normal load. Also, the ball bearings operated at rated conditions with approximately one-third of the predicted requirement. However, it is recommended that coolant flows not fall below 25 gpm for the roller bearings and 50 gpm for each ball bearing for rated speed and load conditions.

Rapid acceleration of the turbopump at startup requires bearing operation with low coolant flow. Acceleration of the 110 mm ball bearing from zero to 13,300 rpm in 1 sec was accomplished with a coolant flow of 20 gpm.

Copper plating the ball bearing raceways to define the wear path should be discontinued because the copper tended to flake off under cryogenic operating conditions.

APPENDIX A

BEARING TEST RESULTS SUMMARY

APPENDIX A

BEARING TEST RESULTS SUMMARY (Sheet 1 of 6)

PHASE I - EVALUATION OF TURBINE END (120 MM) RADIAL BEARINGS

BUILDUP NO.	NO. OF TESTS	PART NO.	SHAFT SPEED RPM	RADIAL LOAD/BRG LB	TOTAL DURATION SEC	COOLANT FLUID	COOLANT FLOW RATE GPM/BRG	COOLANT PRESS PSIG	CONFIGURATION DESCRIPTION	RESULTS
CATEGORY A - SINGLE RADIAL BEARINGS, CONSTANT SPEED - MOTOR-DRIVEN TESTER (FIGURES 3 AND 5)										
1.	8 *	BOWER** KA-1024-EJJ	13,300	11,000	1392	LH_2	35	300	New Bearings	Inboard bearings had uniform roller path, no signs of distress. Outboard bearings had heavily loaded roller path, heavy roller end wear.
	8 *	KA-1024-EJJ	13,300	11,000	4488	LH_2	60	335		
2	24 *	288340	13,200	15,500	5,832	LH_2	20	350	New Bearings	All bearings had inner and outer race wear, roller end wear, cage pocket wear, asperity welding, chips of the guiding shoulder of the outer race, roller skidding.
	60 *	288340	13,300	15,500	12,696	LH_2	30	350		
	8 *	288340	13,300	15,500	2,544	LH_2	20	360		
CATEGORY B - SINGLE RADIAL BEARINGS, ACCELERATION EVALUATION - TURBINE-DRIVEN TESTER (FIGURES 10 AND 11)										
			ACCEL. RPM							
3	12	288340	28,000 13,300	1000	1	LH_2	25	350	New Bearings	Slight roller end wear, minor scratches on roller diameter.
4	5	288340	20,000 13,300	100	1	LH_2	25	350	Same Bearings as B/U 3	Slight roller end wear, no skidding.
5	5	288340	25,000 13,300	500	1	LH_2	25	350	Same Bearings as B/U 3	Slight roller end wear, no skidding.
CATEGORY C - SINGLE RADIAL BEARINGS WITH INITIAL TURBOPUMP CONDITIONS (LOW SPEED, LOW COOLANT FLOW AND PRESSURE AT START) - TURBINE-DRIVEN TESTER (FIGURES 10 AND 11)										
6	1	288340	3300	1500	480	LH_2	25	15	New Bearings	Non uniform race wear path indicating cocked roller. Slight roller end wear, minor cage pocket burning.
	1	288340	5000	3000	360	LH_2	40	45		
	1	288340	9900	4500	162	LH_2	75	95		

* Four single bearings per buildup
** Same as P/N 288340

APPENDIX A

BEARING TEST RESULTS SUMMARY (Sheet 2 of 6)

PHASE II - EVALUATION OF (110 MM) THRUST BEARINGS

BUILDUP NO.	NO. OF TESTS	PART NO.	SHAFT SPEED RPM	THRUST LOAD/BRG LB	TOTAL DURATION SEC	COOLANT FLUID	COOLANT FLOW RATE GPM/BRG	COOLANT PRESS PSIG	CONFIGURATION DESCRIPTION	RESULTS
CATEGORY A - SINGLE THRUST BEARING, CONSTANT SPEED - TURBINE-DRIVEN TESTER (FIGURES 8 AND 9)										
1	3	288410*	13,500	5,000	570	LH_2	38	200	New Bearings	Visual inspection of bearings showed no defects.
	3	288410*	13,500	15,000	570	LH_2	38	200		
	1	288410*	17,000	5,000	252	LH_2	40	190		
	1	288410*	17,000	15,000	252	LH_2	40	190		
2	1	288410*	1,800	4,000	72	LH_2	30	90	Same Bearing as B/U 1	
	1	288410*	1,800	20,000	72	LH_2	30	90	New Bearing	
	1	288410*	1,800	4,000	156	LH_2	45	310	Same Bearing as B/U 1	
	1	288410*	1,800	20,000	156	LH_2	45	310	Same Bearing as Test 2	
	2	288410*	14,000	5,000	156	LH_2	30	310	Same Bearing as B/U 1	
	2	288410*	14,000	26,000	156	LH_2	45	310	Same Bearing as Test 2	
CATEGORY B - SINGLE THRUST BEARING, ACCELERATION EVALUATION - TURBINE-DRIVEN TESTER (FIGURES 10 AND 11)										
			ACCEL.RPM/SEC / RPM							
2	5	288410*	6,000 18,000 / 6,000	5,000	20	LH_2	35	290	Same Bearings as B/U 1 and 2	Evaluation of test parameters indicated good operation.
	5	288410*	18,000 / 4,000	25,000	20	LH_2	35	290	Same Bearings as B/U 2	
	3	288410*	17,000 / 4,000	5,000	12	LH_2	30	320	Same Bearings as B/U 1	
	3	288410*	17,000 / 11,000	25,000	12	LH_2	30	320	Same Bearings as B/U 2	

*1/3 of tandem set

APPENDIX A

BEARING TEST RESULTS SUMMARY (Sheet 3 of 6)

PHASE II - EVALUATION OF (110 MM) THRUST BEARINGS

BUILDUP NO.	NO. OF TESTS	PART NO.	SHAFT SPEED RPM	THRUST LOAD/BRG LB	TOTAL DURATION SEC	COOLANT FLUID	COOLANT FLOW RATE GPM/BRG	COOLANT PRESS PSIG	CONFIGURATION DESCRIPTION	RESULTS
CATEGORY B - SINGLE THRUST BEARING, ACCELERATION EVALUATION (cont'd)										
2	13	288410*	16,000 / 15,000	1,000	18	LH_2	20	300	Same Bearings as Sheet 2 of 6	Examination showed no sign of distress or failure. The load tracts indicate that the load was confined to the design shoulder height. The ball and cage were in excellent condition. One bearing showed blistering of copper plating on the inner race.
	13	288410*	16,000	5,000	18	LH_2	20	300	Same Bearings as Sheet 2 of 6	

*1/3 of tandem set

APPENDIX A

BEARING TEST RESULTS SUMMARY (Sheet 4 of 6)

PHASE II - EVALUATION OF (110 MM) THRUST BEARINGS

BUILDUP NO.	NO. OF TESTS	PART NO.	SHAFT SPEED RPM	THRUST LOAD/SET LB	TOTAL DURATION SEC	COOLANT FLUID	COOLANT FLOW RATE GPM/BRG	COOLANT PRESS PSIG	CONFIGURATION DESCRIPTION	RESULTS
CATEGORY C - TANDEM THRUST BEARINGS, CONSTANT SPEED - MOTOR-DRIVEN TESTER (FIGURES 4 and 5)										
3	6*	288410	13,300	36,000	1020	LH_2	50	350	New Bearings	One bearing set was in excellent condition. The other showed extreme wear with the inner race broken with evidence of excessive heat. The balls were covered with metal deposits. Failure was attributable to lack of coolant due to ice in the supply passages. Flaking of copper plating from the races was evident.
	8*	288410	13,600	5,200	80	LH_2	50	350		
	10*	288410	13,200	40,000	1666	LH_2	50	380		
4	2*	288410	14,200	7,000	12	LH_2	50	350	New Bearings	Misalignment of bearing and spray ring caused rubbing of inner race and spray ring and failure.
CATEGORY D - TANDEM THRUST BEARINGS - INITIAL TURBOPUMP CONDITIONS (LOW SPEED, LOW COOLANT FLOW AND PRESSURE AT START - MOTOR-DRIVEN TESTER (FIGURES 4 AND 5)										
5	2*	288410	3300	15,000	1200	LH_2	13	75	One New, One Reused from B/U 4	
	2*	288410	6600	30,000	400	LH_2	25	135		
6	2*	288410	9900	30,000	60	LH_2	100**	235	One New, One Reused (New on B/U 6)	
7	4*	288410	5000	25,000	252	LH_2	--	90	One New, One Reused	The rolling contact surfaces of the inner and outer race of one bearing were heavily pitted, indicating high loading. This bearing is not usable for further testing.
	2	288410	9700	15,000	145	LH_2	--	230		
	6	288410	9900	30,000	266	LH_2	--	235		

* Two Tandem Sets per Buildup
**Coolant flow decreased to negligible amount because of flow control malfunction during run.

APPENDIX A

BEARING TEST RESULTS SUMMARY (Sheet 5 of 6)

PHASE II - EVALUATION OF (110 MM) THRUST BEARINGS

BUILDUP NO.	NO. OF TESTS	PART NO.	SHAFT SPEED RPM	THRUST LOAD/BRG LB	TOTAL DURATION SEC	COOLANT			CONFIGURATION DESCRIPTION	RESULTS
						FLUID	FLOW RATE GPM/BRG	PRESS PSIG		
CATEGORY E - SINGLE THRUST BEARINGS, CONSTANT SPEED, REVERSED LOAD - MOTOR-DRIVEN TESTER (FIGURES 4 AND 5)										
8	1	288410	6000	9,500	132	LH_2	35	90	Same as B/U 7	Normal Wear, discoloration of balls and race
	2	288410	6000	15,000	162	LH_2	35	90		
9	1	288410	6000	0	234	LH_2	30	95	One Bearing from B/U 5 One Bearing from B/U 7	Normal Wear.

APPENDIX A

BEARING TEST RESULTS SUMMARY (Sheet 6 of 6)

PHASE III - EVALUATION OF PUMP END (110 MM) RADIAL BEARINGS

BUILDUP NO.	NO. OF TESTS	PART NO.	SHAFT SPEED RPM	RADIAL LOAD/BRG LB	TOTAL DURATION SEC	COOLANT FLUID	COOLANT FLOW RATE GPM/BRG	COOLANT PRESS PSIG	CONFIGURATION DESCRIPTION	RESULTS
CATEGORY A - SINGLE RADIAL BEARINGS, WITH CONSTANT SPEED - MOTOR-DRIVEN TESTER (FIGURE 4)										
1.	6*	288260	13,300	5000	1020	LH_2	50	350	New Bearings	
	8*	288260	13,600	1000	80	LH_2	50	400		
	10*	288260	13,200	5000	1670	LH_2	50	360		
2	2*	288260	14,200	8000	20	LH_2	50	350	New Bearings	Excellent Condition
CATEGORY B - SINGLE RADIAL BEARINGS, INITIAL TURBOPUMP CONDITIONS (LOW SPEED, LOW COOLANT FLOW AND PRESSURE AT START) - MOTOR-DRIVEN TESTER (FIGURE 4)										
3	2*	288260	3300	1000	1200	LH_2	13	75	Same Bearings as B/U 2	
	2*	288260	6600	1000	400	LH_2	25	135		
4	2	288260	9900	1000	60	LH_2	100	235	Same Bearings as B/U 2	
5	4	288260	5000	1000	250	LH_2	--	90	Same Bearings as B/U 2	
	8	288260	9900	1000	528	LH_2	--	235		
6	6	288260	6000	1000	588	LH_2	40	90	Same Bearings as B/U 2	
7	2	288260	6000	1000	470	LH_2	40	95	Same Bearings as B/U 2	Excellent Condition

* Two Single Bearings per Buildup